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COMPRESSOR NOISE REDUCTION ON A REFRIGERATOR

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ABSTRACT

The need of refrigerators and freezers, both commercial and domestic, without "acoustic pollution" is getting more and more strong, particularly in the European market.

In many flats, the rooms are disposed so that a noisy freezer situated in the kitchen, can be heard in all other rooms causing a strong disturbance particularly during the rest hours.

In this paper we present a study carried out to reduce the noise of a compressor installed on an appliance, beginning from the definition of a new housing.

1. CURRENT COMPRESSOR NOISE CHARACTERISTICS

The compressor involved in this study is a 1/5 HP, low back pressure, high efficiency, R134a, reciprocating type.

The average sound power spectrum of present compressors as measured in a semi-anechoic room is reported in Fig.1.

In Fig.2 is reported the noise spectrum of the compressor as measured on a domestic freezer (220 lt volume).

As shown in the spectra, the contribution to the noise is given by:

a) Peaks in the region up to 315 Hz

b) Peaks in the 500 Hz region

c) Peaks in the 2000 Hz region

The first region noise is affected by discharge pulsation and by compressor vibration.

The second region noise is related to the suction cavity excitation and the third one is radiated by the shell vibrating on its modal frequencies, excited by the internal noise sources.

A very simple flow chart of the noise generation and transmission path is reported in Fig.3.

2. NOISE IMPROVEMENTS

2.1 1st Step: Shell redesign

To reduce the noise radiated by the excited shell, the following design modifications were analyzed:

- Increase the overall stiffness to raise the resonant frequency and reduce the vibration amplitudes

- Avoid abrupt changes in shell curvature that act as semi-rigid boundary conditions

- Move the suspension springs support to locations of high stiffness, where the input mobility of the shell is reduced; the transmission of vibration energy to the shell is therefore minimized

- Eliminate, as much was possible, flat surfaces

The adoption of a new redesigned housing has then been evaluated as the best solution to eliminate the drawbacks of the compressor in terms of noise and vibrations transmitted.

The basic strategy we have pursued was the elimination, as much was possible, of the flat surfaces.

Another important aspect we wanted to point out was the modal density; it was advisable the resonant frequencies of the shell would be spaced enough to avoid double mode excitation.

The first design changes were concentrated on the reduction of the side flat surfaces; the top and the bottom of the shell were also involved in the design variations to eliminate the abrupt changes at the bending points.
The Finite Element Analysis showed an increase in the modal frequencies, but lower than we expect.
A more heavy design variation was then studied, involving also the overall dimensions of the compressors.
To increase the curvature with the constrain of the pumping mechanism, we were forced to slightly increase the in plan dimension of the shell.
The redesign of the suspension system, allowed us to reduce the compressor height.
The final redesign is showed in Fig.4 together with the present shell and the first design change.

Fig.5a shows the frequencies comparison from Finite Element Analysis.

The experimental evaluation with modal analysis of the new shell in comparison with the present one, confirmed the frequency shift towards higher frequencies as shown in Fig.5b.
The reduction in vibration amplitude was also reached.

2.2 2nd Step: Discharge pulsation reduction

As mentioned before, an important contribution to the refrigerator noise is given by the discharge gas pulsation. It is well known that the presence of harmonics in the pulsation spectrum can excite the high pressure side of the refrigerator.

To "weight" the influence of the pulsation effect on the noise of the compressor installed on the refrigerator, a compressor with the discharge line heavily modified was developed.
An additional volume was inserted on the discharge line to eliminate the high harmonic contents.

The Fig.6 shows the pulsation spectrum of the modified compressor.

3. GAS CAVITY

As the new shell has different dimensions in comparison with the present one, a different cavity frequency was expected. The housing design was also evaluated considering the gas cavity excitation in the 400-630 Hz region, typical for the noise due to the cavity excitation.

The cavity frequencies are related to the refrigerant conditions in the shell; during the running period of the compressor on the refrigerators, both suction pressure and compressor temperatures change continuously.
According to the data recorded from different refrigerators and freezers, the shell temperature usually ranges from 50 to 70°C; we assumed as suction pressure the standard rating point (i.e. -23.33°C).

The experimental evaluation of the cavity noise showed that the cavity excitation occurs at running conditions different from the typical ones in the refrigerator as shown in Fig.8.

4. NOISE VERIFICATION ON APPLIANCE

A set of compressors assembled with the new shell and the modified discharge line was tested in a semi-anechoic room.

A reduction of 5 dB in sound power level was reached, with the expected variation in the noise spectrum. In Fig.8 is shown the noise spectrum of the modified compressors in comparison with the present one.
The test was repeated with the compressor installed on a domestic freezer (220 lt volume) and the noise reduction was confirmed as shown in Fig.9.

5. CONCLUSION

Redesign of the shells increasing their stiffness and shifting their natural frequencies towards higher values, has been confirmed a powerfull method to reduce the compressor noise. Considering the compressor installed on a refrigerator, it must be avoided the excitation of the discharge tubing and the condensers, optimizing the discharge pulsation.

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- Soedel W. - "Vibration of shells and plates" - M.Dekker

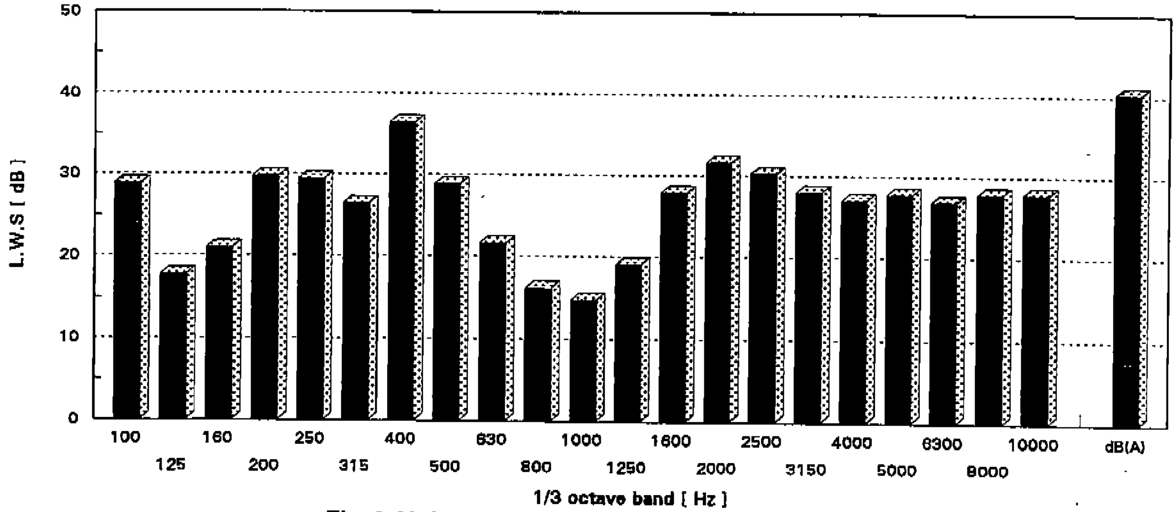


Fig.1 Noise spectrum of present compressor

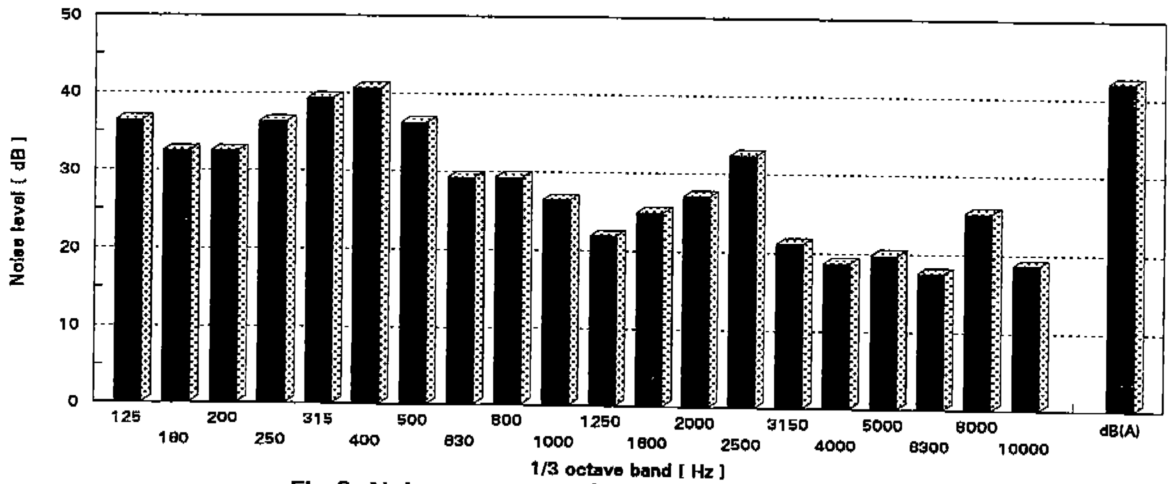


Fig.2 Noise spectrum of present compressor on appliance

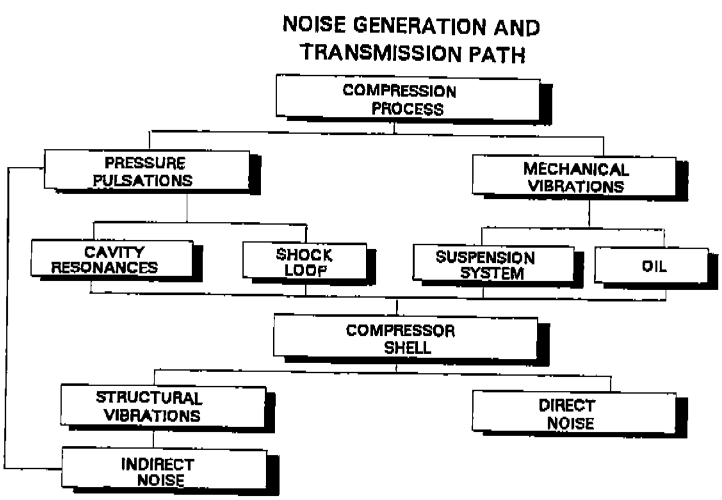
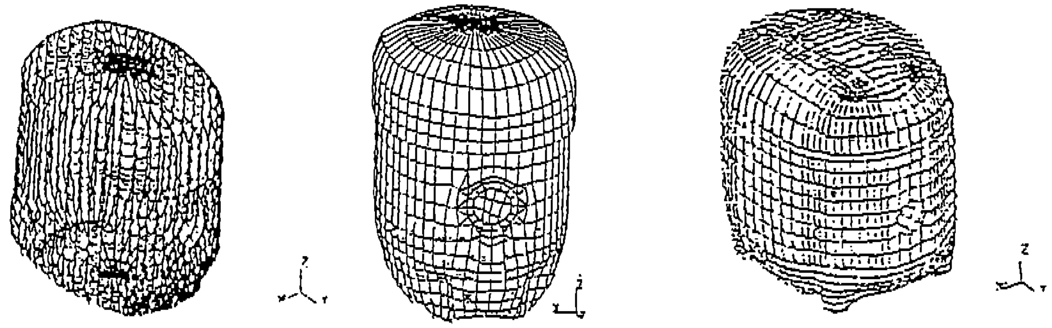


Fig.3



a) present shell b) first design c) final design

fig.4 Development shell design

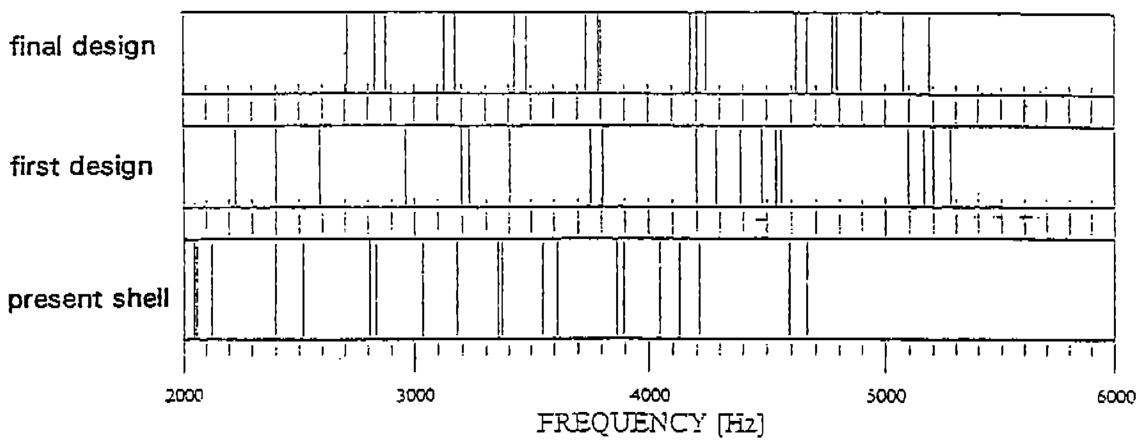


fig.5a Theoretical evaluation

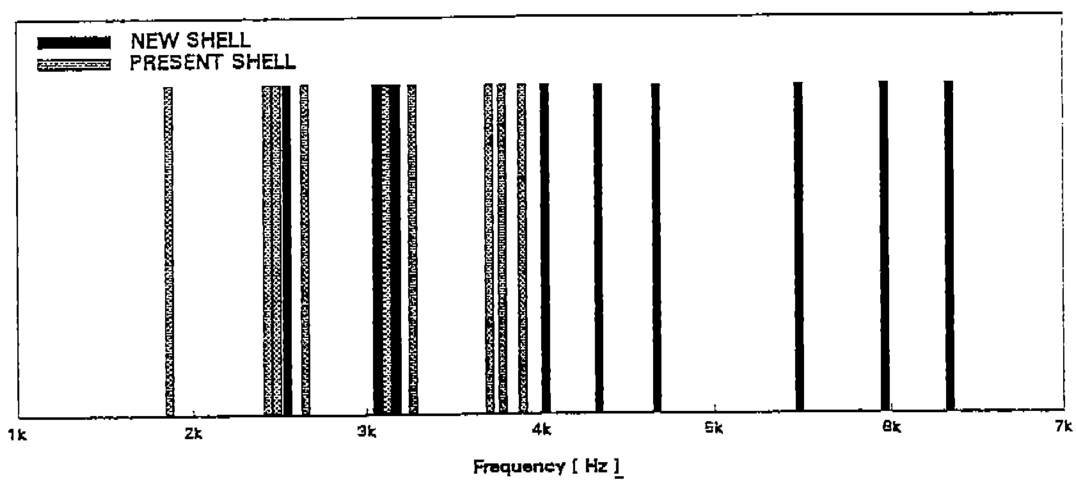


fig.5b Modal analysis evaluation

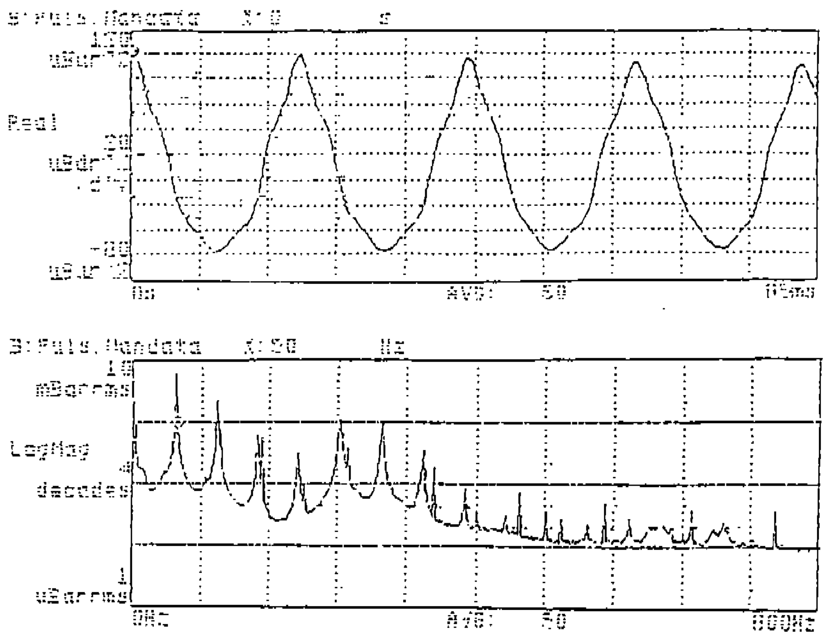


Fig.6 Discharge gas pulsation

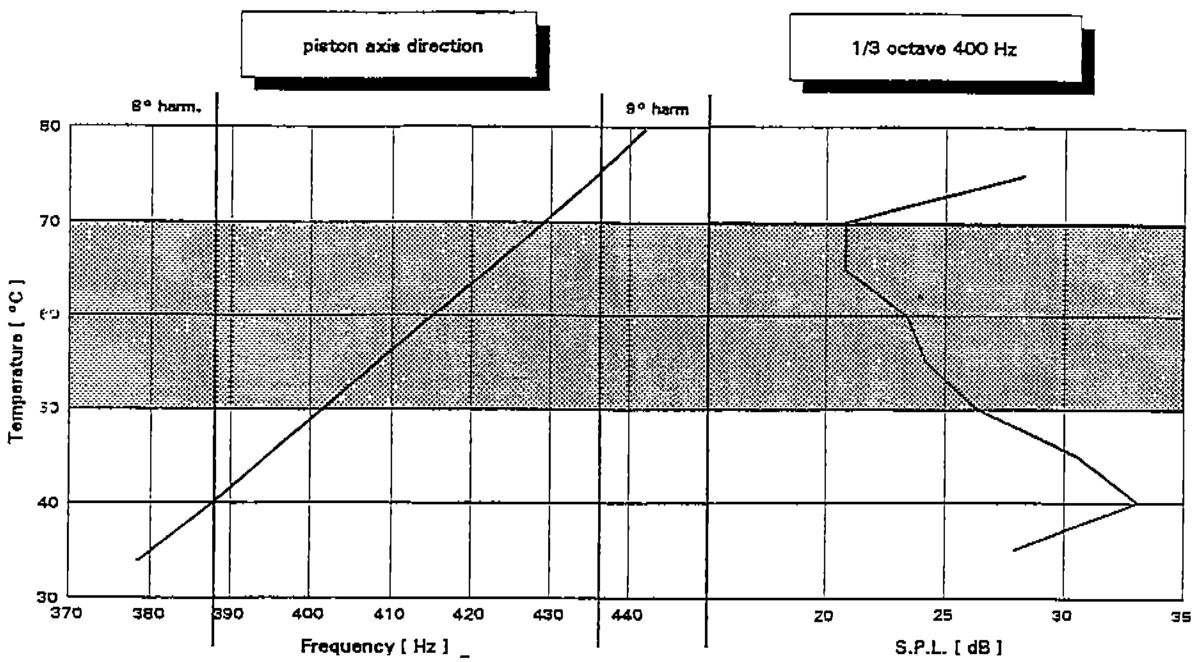


Fig.7 Experimental gas cavity identification

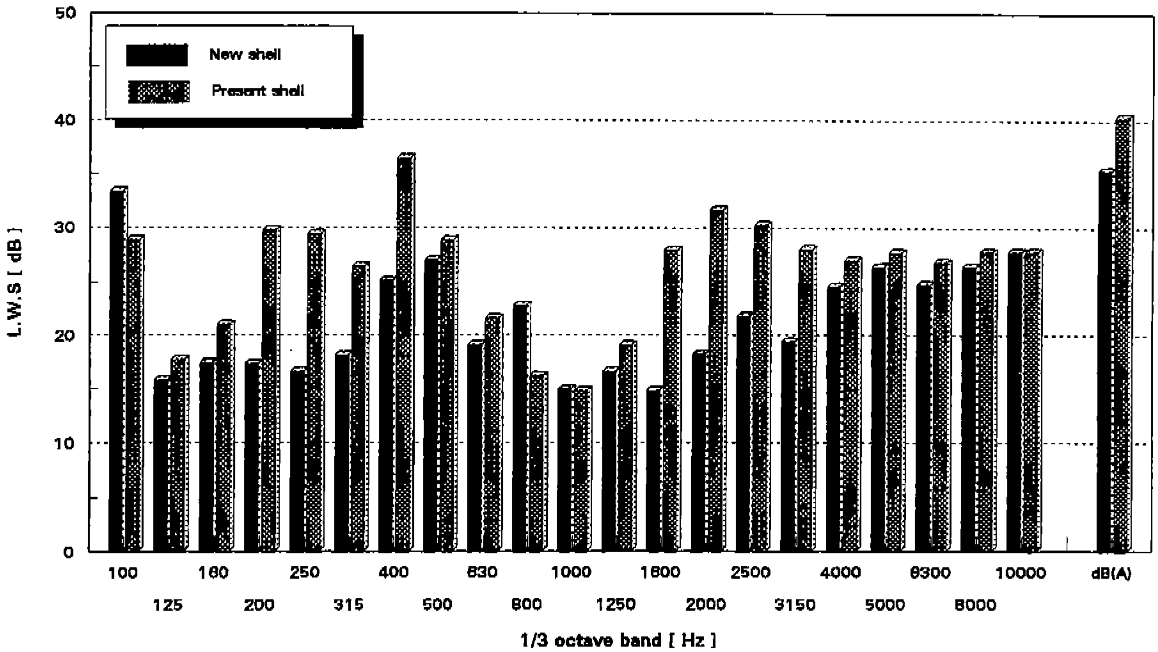


Fig.8 Noise spectrum of modified compressor vs. present compressor

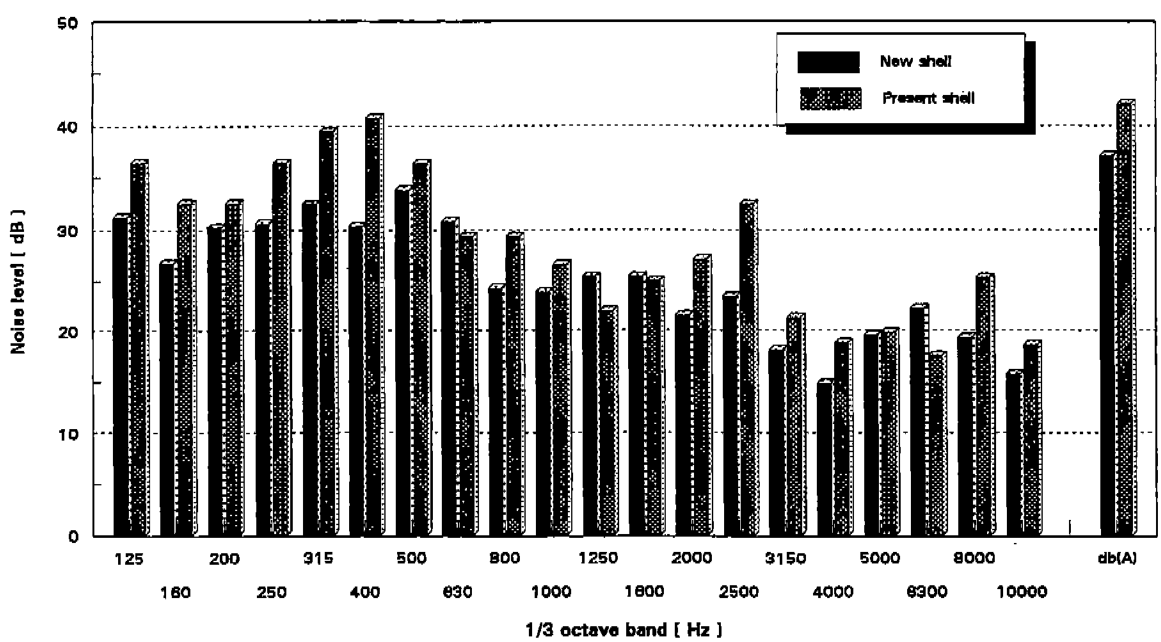


Fig.9 Appliance test